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# An Evaluation of Superheat-Based Refrigerant Charge Diagnostics for Residential Cooling Systems

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## ABSTRACT

*Although refrigerant charge has an important influence on the performance of residential cooling systems with fixed orifice metering devices, there has been little research to quantify the effects of incorrect charge or design new diagnostics for evaluating charge level. The most common diagnostic for charge level in these systems is the superheat test. In this paper, we examine three superheat technologies/techniques. Two of the diagnostics are appropriate for detecting incorrect charge; one is not. Additionally, measurements at four houses indicate that it is important to measure the condenser air entering temperature with a high degree of accuracy. Measurement of the wet bulb temperature in the return plenum and suction line temperature are equally important, but seemingly easier then measuring the condenser air temperature, as several measurement technologies yielded similar results for these quantities. The importance of refrigerant charge to energy use and capacity of residential cooling systems, the limitations of the superheat test, and the variations in the test method results and interfaces necessitate the development of a standard method or methods to determine refrigerant charge level.*

## INTRODUCTION

Based on tests of more than 4000 residential cooling systems in California, it is clear that many systems have incorrect refrigerant charge levels (Proctor 2000). Data from these tests indicate that about 34% are undercharged, 28% are overcharged, and only 38% have correct charge. In the past, data from Blasnik et al. (1996) and Proctor (1997, 1998) have indicated that an undercharge of 15% is common.

Both undercharge and overcharge can reduce cooling equipment longevity, capacity, and efficiency. For example, laboratory test data for capillary-tube-controlled equipment (Farzad and O'Neal 1988) indicate that an undercharge of 15% will reduce cooling equipment total capacity by 8 to 22% and its energy efficiency ratio (EER) by 4 to 16%. An overcharge of 10% will reduce capacity by 1 to 9% and EER by 4 to 11%. Figures 1 and 2 show the capacity degradation of a 3 ton split-system air conditioner versus charge for various outdoor temperatures. These two figures indicate that TXV-controlled equipment is much less sensitive to deviations from the correct charge (Farzad and O'Neal 1989). Their work showed a similar pattern for EER degradation.

Laboratory test data indicate that some short-tube-orifice-controlled equipment behaves more like TXV-controlled equipment (O'Neal et al. 1989), while others behave like capillary-tube-controlled equipment (Rodriguez 1995). Additional research with a larger sample of short-tube-orifice-controlled cooling equipment is needed to clarify how the performance of equipment like this depends on charge.

Several diagnostics are available to assess the correct amount of refrigerant charge in a system, but only the superheat and subcooling tests are practical, well developed, and reliable (when properly done). Superheat tests are for capillary-tube-controlled equipment; subcooling tests are for TXV-controlled equipment. The common assumption is that superheat tests are also appropriate for short-tube-orifice-controlled equipment. Neither test is standardized, but equipment manufacturers commonly specify them. However, many service technicians do not use the tests, primarily because of the time it takes to do them, but also because they do not have necessary equipment or appropriate indoor and outdoor conditions for the test.

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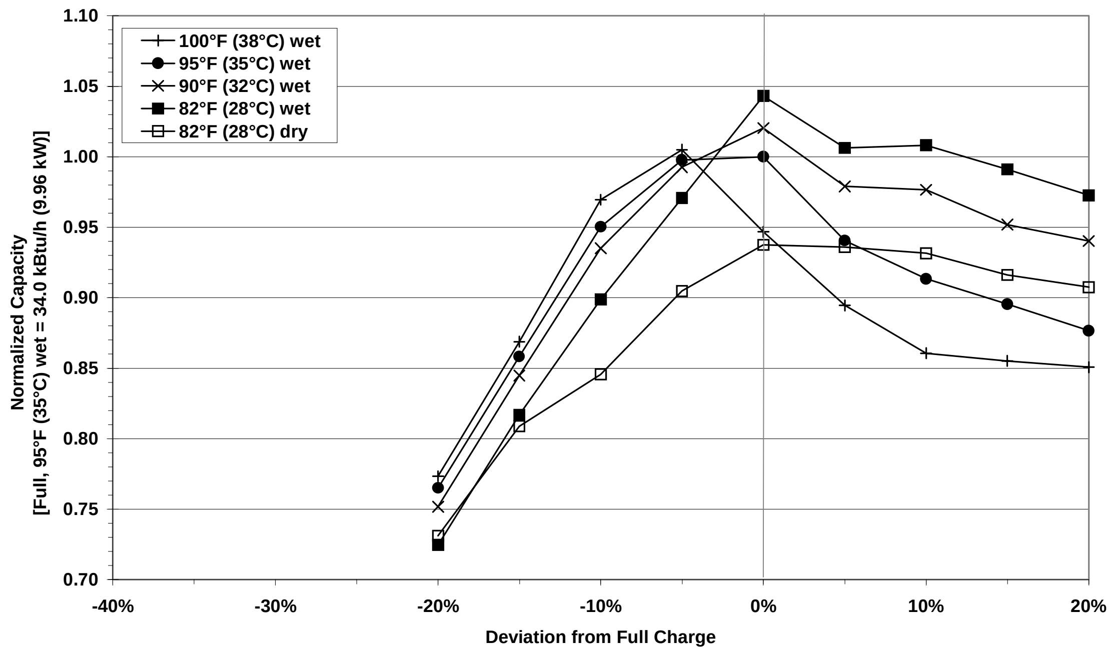


**Figure 1: Total Capacity Variation – Capillary Tube (Farzad and O'Neal 1988)**

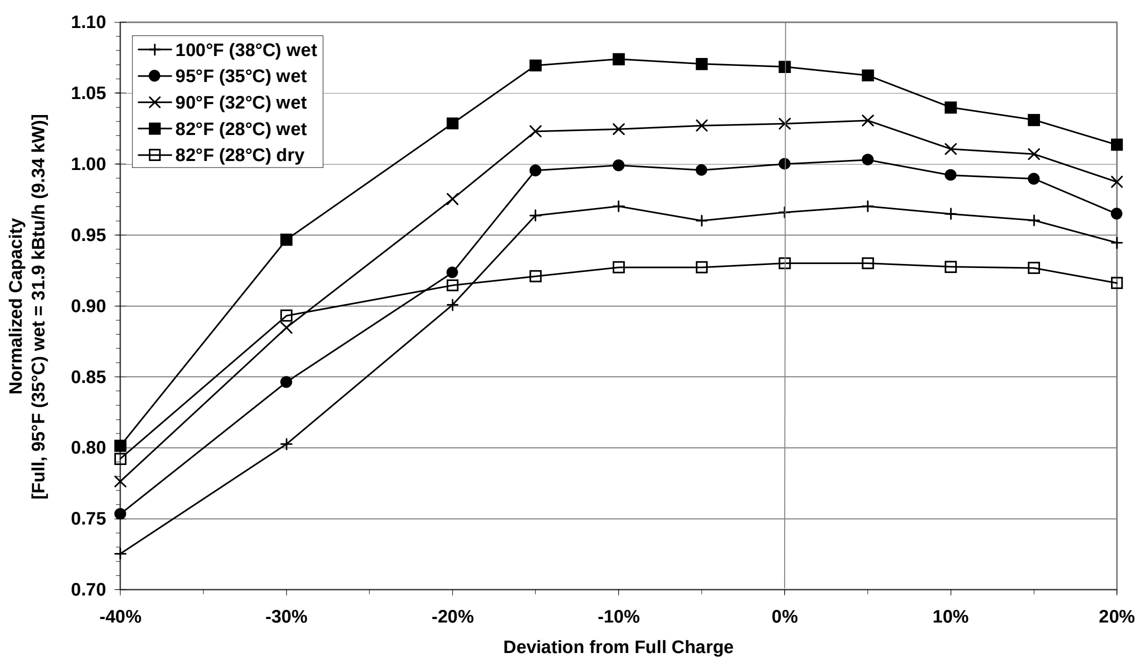


**Figure 2: Total Capacity Variation – TXV (Farzad and O'Neal 1989)**

## SUPERHEAT TEST METHODS EVALUATED

Carrying out superheat tests on systems with a capillary tube or short-tube orifice is more important than subcooling tests on TXV equipped systems, because a TXV tends to mitigate charge deficiency effects. Consequently, we focused our evaluation on three state-of-the art methods that may facilitate superheat tests on residential cooling systems. Each method involves different hardware, software, and measurements. All three methods, which we broadly characterize as "Superheat Calculation Methods", make the superheat test easier. Table 1 characterizes the methods, as well as our data collection truth standard (Reference). Although the methods include other tests (e.g., Method 3 also assesses air-handler airflow), our evaluation only focuses on the ability of the methods to assess refrigerant charge.

**Table 1: Superheat Test Methods Evaluated**

| Method | Description |
|---|---|
| Reference | Uses Method 3 software (see below) with continuously monitored data collected using a research-grade data acquisition system. This method is our "Truth Standard" for purposes of comparison. |
| 1 | Personal Digital Assistant (PDA) with attached refrigerant pressure transducers and temperature sensors, which is intended for use on light-commercial buildings. PDA also runs software that provides diagnoses in the field. The data can be uploaded later from the PDA to a web site that provides a more detailed analysis. |
| 2 | Data collection system including pressure transducers, temperature and humidity sensors, software, and computer that collects data for superheat test and then provides recommended action to technician. Currently designed for use by an authorized crew who carries out the test, provides diagnoses to a contractor, and then verifies the efficacy of any work that is performed. |
| 3 | Software program that requires the technician to enter single point data from refrigerant manifold gauges and temperature sensors into a computer, which then provides diagnoses based on measured and target superheat comparison. To avoid computer use in the field, the contractor also has the option of providing input data by telephone to a remote operator, who then enters it into the software and relays diagnoses back to the technician. |

Our goal in the evaluations was to assess the accuracy of these methods and to ascertain whether they are useful for residential commissioning. The central questions we are attempting to answer with this work are:

- What technologies are available to make a superheat test easier?
- How do these technologies work?
- How accurate are they, and how accurate do they need to be to accurately assess charge?

There are three caveats regarding our evaluation:

1. Our evaluation explored the strengths and limitations of the methods that we tested rather than testing them on a statistically significant number of homes. In particular, we only tested cooling equipment in four houses (two new ones and two older ones), all with short-tube-orifice controls.
2. Although we ostensibly compare different methods that might facilitate the superheat test, we could not always do an "apples-to-apples" comparison. The different methods all use different input data and different algorithms. These algorithms are either proprietary or were not accessible to us. Consequently, our results describe differences between the different methods and we state when the quantities being compared are not effectively the same.
3. As a reference standard, we used the superheat test even though it is prone to measurement error, can be time consuming, and the indoor and outdoor conditions that govern its use are limited. In particular, our evaluations used continuously monitored data that we collected and output from the software of Method 3 based on these data. We chose that method because it is the most fully developed product that we considered and because it has an extensive history of field tests. We also chose it because we did not have time or resources to do gravimetric tests, the preferred truth standard. A gravimetric analysis is useful for a

wider range of conditions and is potentially more accurate. That method involves removing all of the refrigerant in a system, drawing a vacuum and leak testing the system, and then adding the manufacturer's recommended amount of refrigerant for the compressor, coils, and installed "line set" (refrigerant line length). However, the gravimetric test requires a matched indoor coil and outdoor unit, as well as a good measurement of the line set. Further drawbacks are the time and skill that it takes to do a gravimetric test, which is more complex than a superheat test.

## WHAT IS THE SUPERHEAT TEST?

To understand our evaluations, it is important to understand the basic theory of the superheat test. Superheat is a thermodynamic metric, defined as the temperature rise above the vapor saturation temperature (i.e., the temperature at which all the liquid in a mixture is evaporated for a given pressure). For typical suction line pressures of 40 to 100 psig (276 to 690 kPa) found in R-22 systems, the vapor saturation temperature will range from 17 to 59ºF (-8 to 15°C). Refrigerant pressure gauges commonly have a concentric scale to conveniently indicate the vapor saturation temperature that corresponds to the measured pressure for a given refrigerant.

For a superheat test of a cooling system, the actual superheat at operating conditions is determined by measuring the temperature and pressure of the refrigerant in the suction line, just before the refrigerant enters the compressor. The actual superheat is the difference between the suction line refrigerant temperature and the vapor saturation temperature at the measured pressure. The deviation of this superheat value from a target superheat is an indicator of whether the amount of refrigerant charge in the air conditioner is correct when it operates at design conditions. Note that the surface temperature of the suction line rather than the refrigerant temperature itself is measured in practice. This difference is a potential source of error in the test, as discussed later when we review our field test results.

Equipment manufacturers use laboratory tests to determine target superheat values as a function of the return air wet-bulb temperature entering the evaporator and of the outdoor air dry-bulb temperature entering the condenser. Those temperatures act as surrogate metrics to characterize two variables that affect the refrigerant evaporation rate in the evaporator: the cooling load on the evaporator and the heat rejection rate of the condensing unit. These data are typically available in a table or in a chart. In a superheat test, the technician measures the two air temperatures that we describe above and then obtains the corresponding target superheat from the table or chart.

If the measured superheat value is too low compared to the target superheat, there will be too much charge in the system at design operating conditions. In addition to energy and capacity impacts, there is a chance that liquid refrigerant will not completely evaporate at these conditions and could slug the compressor. If the measured superheat value is too large compared to the target superheat, there will be too little charge in the system at design operating conditions. In addition to energy and capacity impacts that are more serious than in the overcharge case, the suction line pressure and corresponding saturation temperature of the refrigerant will be very low. This can lead to ice formation on the evaporator, which can restrict heat transfer, increase airflow resistance, and reduce air-handler airflow. This will further reduce air conditioner performance and can shorten compressor life.

## TEST HOUSES AND COOLING EQUIPMENT

We used each of the methods to test four separate California houses, which are cooled by split-system central air conditioners equipped with short-tube-orifice metering devices and R-22 refrigerant. The rated capacity of all the tested air conditioners is 3 to 4 tons. Table 2 summarizes relevant house and equipment characteristics.

**Table 2: Test House and Cooling Equipment Characteristics**

| Site | House Location | Cooling Equipment Age [Years] | Condensing Unit Rated Capacity [Tons (kW)] | Evaporator Rated Capacity [Tons (kW)] |
|---|---|---|---|---|
| A | Larkspur, CA | 17 | 3 (11) | Unknown |
| B | Sacramento, CA | < 1 | 3.5 (12) | 4 (14) |
| C | Sacramento, CA | < 1 | 3 (11) | 4 (14) |
| D | Concord, CA | > 15[1] | 3.5 (12) | Unknown |

[1]Estimated. The house was about 25 years old and was occupied by the current owner 4 years ago. The system is very old and decrepit.

We conducted superheat tests in the "as found" condition and then repeated them each time after we added or removed refrigerant charge. However, we only report the measurements for the "as found" condition and the final "post-charging" condition. At each house, we operated the equipment for at least 15 minutes initially and after each charge change to allow system conditions to stabilize before conducting a superheat test.

## FIELD TEST RESULTS

The following describes our field test results, compares the data that are required to determine the actual and target superheat values, and compares diagnoses generated by the various methods. In many cases, some of the field data are missing. Most of the time, one should interpret this as a failure of a product or operator error, indicated by "NA" in the tables. There are a couple of exceptions, indicated with "-" in the tables: we did not use Method 3 at Site D because the technician was unavailable. Additionally, Method 1 does not measure the return plenum wet-bulb temperature or use it in its analysis. We discuss the details and implications of intentional and unintentional missing data more fully throughout the following sections.

### Determining the Target Superheat

An important part of conducting the superheat test is determining the target superheat. Methods 2 and 3 use a chart for this purpose, similar to the manufacturers chart described earlier. To use the chart, Method 3 requires measuring the dry-bulb temperature of the air entering the condenser and the wet-bulb temperature of the air entering the evaporator from the return plenum. It seems that Method 2 may use additional calculations (discussed in further detail later). Method 1 does not measure either of these temperatures. The manufacturer reports that this method uses a proprietary superheat chart, which ascertains the loads on the coils through some other algorithm. That method requires measuring the ambient temperature at the PDA, which is likely theorized to be close to the condenser entering temperature, because the short length of the device's refrigerant hoses requires it to be located near the condenser. It is not clear what Method 1 uses in place of the return plenum wet-bulb temperature, because it has no temperature or humidity sensors inside the house or ducts.

For each site, Table 3 lists the measured air temperature entering the condenser and the return plenum wet-bulb temperature for the "as found" and "post-charging" conditions.

With the Reference results as a benchmark, it is clear that Method 1 sometimes measures a different temperature than the condenser entering temperature, even when its temperature sensor is no more than 6 feet away from the condensing unit. For example, in the Site D post-charging case, it measures a temperature that is approximately 16°F (9°C) too high. Since it is not clear how the Method 1 algorithm works, it is not clear how this difference might affect the method's analysis. However, it seems likely that any attempt to use this temperature to determine the load on the condenser might lead to significant errors.

With the exception of Method 1 at Site D, all methods do a reasonable job measuring the condenser air entering temperature. However, there are still substantial differences in some cases (up to 4°F (2°C), Site B post-charging). Also, there is a noticeable deviation between methods at Site A. That deviation is difficult to analyze because the Reference condenser entering temperature sensor failed at this site. The other deviations do not exceed 2°F (1°C).

**Table 3a: Measured Parameters for Determining "As Found" Target Superheat**

| Method | Condenser Air Entering Temperature [°F (°C)] | | | | Return Plenum Wet-Bulb Temperature [°F (°C)] | | | |
|---|---|---|---|---|---|---|---|---|
| | Site A | Site B | Site C | Site D | Site A | Site B | Site C | Site D |
| Ref. | 96[1] (36) | 85 (29) | 89 (32) | 76 (24) | 65 (18) | 62 (17) | 66 (19) | 62 (17) |
| 1 | 105 (41) | 88 (31) | 87 (31) | 85 (29) | - | - | - | - |
| 2[2] | 108 (42) | 85 (29) | 91 (33) | 73 (23) | 66 (19) | 61 (16) | 65 (18) | 61 (16) |
| 3 | 100 (38) | 85 (29) | NA[3] | - | 67 (19) | 63 (17) | NA[3] | - |

[1]This value is incorrect – it is the ambient air temperature. The temperature probes measuring condenser entering air temperature failed for this test.
[2]Method 2 measures the humidity and temperature at the return grille(s). The number reported here is the average return grille wet-bulb temperature and not the return plenum wet-bulb temperature.
[3]Either operator error or a computer error caused the Method 3 data to be missing for Site C as found.

**Table 3b: Measured Parameters for Determining "Post Charging" Target Superheat**

| Method | Condenser Air Entering Temperature [°F (°C)] | | | | Return Plenum Wet-Bulb Temperature [°F (°C)] | | | |
|---|---|---|---|---|---|---|---|---|
| | Site A | Site B | Site C | Site D | Site A | Site B | Site C | Site D |
| Ref. | 100[1](38) | 88 (31) | 92 (33) | 78 (26) | 66 (19) | 61 (16) | 64 (18) | 60 (16) |
| 1 | 98 (37) | 89 (32) | 92 (33) | 94 (34) | - | - | - | - |
| 2[2] | 105 (41) | 92 (33) | NA[3] | 77 (25) | 65 (18) | 60 (16) | NA[3] | 60 (16) |
| 3 | 101 (38) | NA[4] | 92 (33) | - | 66 (19) | NA[4] | 65 (18) | - |

[1]This value is incorrect – it is the ambient air temperature. The temperature probes measuring condenser entering air temperature failed for this test.
[2]Method 2 measures the humidity and temperature at the return grille(s). The number reported here is the average return grille wet-bulb temperature and not the return plenum wet-bulb temperature.
[3]A computer error caused the Method 2 data to be missing for Site C post-charging.
[4]Either operator error or a computer error caused the Method 3 data to be missing for Site B post-charging.

The other measurement necessary to determine the target superheat is the wet-bulb temperature of air entering or that will enter the evaporator. There are major differences between methods in how they measure this quantity:

- Reference uses a thermistor and a relative humidity (RH) sensor in the return plenum.
- Method 1 does not measure this parameter.
- Method 2 uses the average of all RH and temperature measurements at the return grilles.
- Method 3 uses a thermocouple surrounded by a wet cotton sleeve in the return plenum.

Method 2 does not account for the effects of duct leakage, because it measures the temperature at the return grille instead of within the plenum. In systems with substantial return duct leakage such as Sites A and D (17 and 33% respectively), we expect Method 2 will measure a lower wet-bulb temperature, which will result in a target superheat that is lower than it should be. However, there is very little difference in the wet bulb temperatures, no matter what measurement methodology is used. Part of this is because Sites A and D have most of their ducts in a crawlspace and garage respectively, which are buffer spaces with similar temperature and humidity conditions as the house. If the ducts had been located in a hot attic, one would expect a substantial effect. In any case, from this small sample, there seems to be no effective difference between the various measurement techniques for this quantity.

### Determining the Actual Superheat

There is much more consistency with how the three methods measure parameters that determine the actual superheat value. To establish the actual superheat, each method measures the temperature of the suction line near the service port at the compressor and the pressure in the suction line at this service port. The suction line pressure is then converted to a vapor saturation temperature using a standard vapor saturation temperature chart. The difference between the suction line and vapor saturation temperatures is the actual superheat. Table 4 lists the measured suction line pressures and Table 5 lists the associated saturation temperatures and the measured suction line temperature.

Several patterns are clear from these data. Differences in suction line pressures (up to 6 psig (41 kPa) or 11%) are sometimes greater than we expect based on the accuracy that is achievable for the types of gauges used. This means that vapor saturation temperatures will also be noticeably different. In the case of the largest pressure difference (Site C as found), there is a corresponding 5°F (3°C) difference in calculated vapor saturation temperatures, which will contribute to a 5°F (3°C) difference in actual superheat. Method 3 uses 5°F (3°C) as the expected accuracy of measurement of actual superheat.

To determine if the variation in pressures between different methods was due to gauge accuracy, we checked the calibration of the pressure gauges and transducers using a Heise pressure gauge (accuracy ± 0.6 psig (4 kPa)) and helium gas at various pressures. There were very small differences of 1 to 4 psig (7 to 27 kPa) between the gauges at typical suction pressures. The larger errors occurred with the Method 2 pressure transducer. The gauges and transducer used in the reference as well as Methods 1 and 3 had errors less than 2 psig (13 kPa), with 1 psig (7 kPa) errors being most common. All gauges exhibited a negative bias.

Given the consistency of pressure gauges during calibration, we do not know why there are pressure differences between methods in the "as found" cases. We hypothesize that measurements at slightly different times under changing operating conditions might be the cause of these pressure differences. Unfortunately, we did not continuously monitor refrigerant pressure. However, periodic fluctuations in suction and liquid line temperatures during the tests seem to provide evidence in support of our hypothesis.

**Table 4a: Measured Pressures for Determining "As Found" Actual Superheat**

| Method | Suction Line Pressure [psig (kPa)] | | | |
|---|---|---|---|---|
| | Site A | Site B | Site C | Site D |
| Ref. | 59 (410) | 58 (400) | 57 (390) | 45 (310) |
| 1 | 63 (430) | 63 (430) | 51 (350) | 49 (340) |
| 2 | 62 (430) | 58 (400) | -56[1] | 45 (310) |
| 3 | 62 (430) | 63 (430) | NA[2] | - |

[1]This value is erroneous and was traced to a computer error in Method 2.
[2]Either operator error or a computer error caused the Method 3 data to be missing for Site C as found.

**Table 4b: Measured Pressures for Determining "Post-Charging" Actual Superheat**

| Method | Suction Line Pressure [psig (kPa)] | | | |
|---|---|---|---|---|
| | Site A | Site B | Site C | Site D |
| Ref. | 67 (460) | 70 (480) | 78 (540) | 64 (440) |
| 1 | 69 (480) | 70 (480) | 78 (540) | 66 (460) |
| 2 | 69 (480) | 67 (460) | NA[1] | 64 (440) |
| 3 | 68 (470) | NA[2] | 78 (540) | - |

[1]A computer error caused the Method 2 data to be missing for Site C post-charging.
[2]Either operator error or a computer error caused the Method 3 data to be missing for Site B post-charging.

In particular, at Site C post-charging, we observed pulsating variations in suction line temperature with an amplitude of as much as 4°F (2°C) with a period of about 2 to 3 minutes peak to peak (see Figure 3). This does not mean that the vapor saturation temperatures (and therefore pressures) varied as much, but it does indicate that they would be varying. One might ask then, if pressures are varying, why are the pressures measured by each method after charge correction identical (Table 5b), and why is there almost no variation over time in suction line temperature before charge correction.

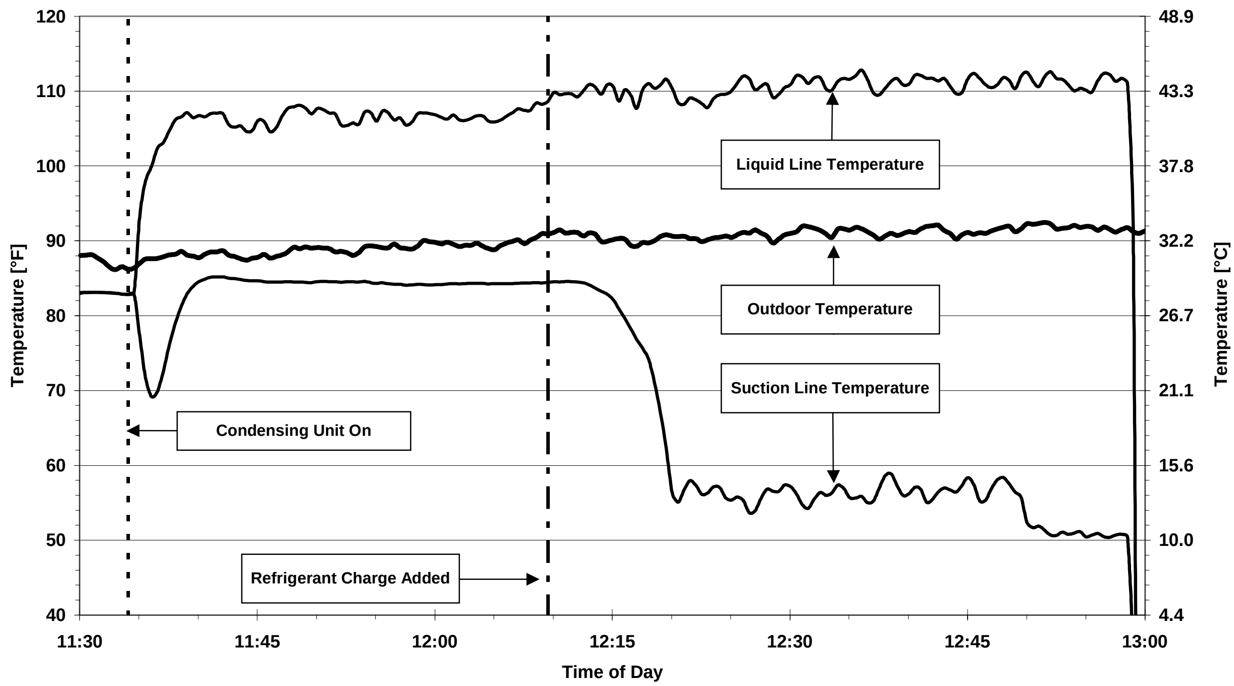


**Figure 3: Site C Field Data**

Regarding the variability in the pressure signal, the agreement between methods might simply be fortuitous, such that the measurements happened to occur at times when the pressures matched. Regarding the relatively constant as-found suction pressure, a possible explanation is that the small temperature differences between the refrigerant vapor and outdoors associated with the undercharge found at this site caused poor heat transfer in the suction line. This tends to mask the effect of pressure fluctuations on suction line temperature. This effect is not masked for the liquid line, because there are substantial temperature differences between the subcooled liquid refrigerant and outdoors. In addition, the heat transfer effectiveness from the subcooled refrigerant to the liquid line tubing wall is greater than that for the superheated refrigerant vapor to the suction line, because of the higher specific heat of the liquid.

**Table 5a: Measured Temperatures for Determining "As Found" Actual Superheat**

| Method | Vapor Saturation Temperature [°F (°C)] | | | | Suction Line Temperature [°F (°C)] | | | |
|---|---|---|---|---|---|---|---|---|
| | Site A | Site B | Site C | Site D | Site A | Site B | Site C | Site D |
| Ref. | 34 (1) | 32 (0) | 32 (0) | 22 (-6) | 73 (23) | 79 (26) | 84 (29) | 68 (20) |
| 1 | 36 (2) | 36 (2) | 27 (-3) | 25 (-4) | 74 (23) | 76 (24) | 84 (29) | 68 (20) |
| 2 | 35 (2) | 32 (0) | -28[1] | 22 (-6) | 65 (18) | 77 (25) | 87 (31) | 68 (20) |
| 3 | 37 (3) | 35 (2) | NA[2] | - | 75 (24) | 76 (24) | NA[2] | - |

[1]This value is erroneous and was traced to a computer error in Method 2.
[2]Either operator error or a computer error caused the Method 3 data to be missing for Site C as found.

**Table 5b: Measured Temperatures for Determining "Post-Charging" Actual Superheat**

| Method | Vapor Saturation Temperature [°F (°C)] | | | | Suction Line Temperature [°F (°C)] | | | |
|---|---|---|---|---|---|---|---|---|
| | Site A | Site B | Site C | Site D | Site A | Site B | Site C | Site D |
| Ref. | 39 (4) | 40 (4) | 47 (8) | 36 (2) | 47 (8) | 45 (7) | 56 (13) | 46 (8) |
| 1 | 40 (4) | 41 (5) | 46 (8) | 39 (4) | 48 (9) | 48 (9) | 59 (15) | 46 (8) |
| 2 | 39 (4) | 39 (4) | NA[1] | 37 (3) | 43 (6) | 44 (7) | NA[1] | 45 (7) |
| 3 | 40 (4) | NA[2] | 47 (8) | - | 48 (9) | NA[2] | 56 (13) | - |

[1]NA = Not Available. A computer error caused the Method 2 data to be missing for Site C post-charging.
[2]Either operator error or a computer error caused the Method 3 data to be missing for Site B post-charging.

The suction line temperatures in Table 5 show about the same amount of variation as the vapor saturation temperatures, but in a different pattern. Unlike the pressure measurements on which the latter temperatures are based, each method uses a different device to measure the suction line temperature and it is a much more complex parameter to measure.

- Reference uses a thermistor that is cable-tied to the suction line and insulated.
- Method 1 uses a large proprietary temperature sensor with an elastic clamping mechanism.
- Method 2 uses an RTD sensor that is cable-tied to the suction line and insulated.
- Method 3 uses a thermocouple that is taped to the suction line and insulated.

The actual quantity needed is the temperature of the refrigerant inside the copper refrigerant line. However, this quantity is not measurable directly, so all methods measure the outside wall temperature of the copper tubing, assuming that the turbulence in the refrigerant and the high conductivity of the copper will transfer heat well from the refrigerant to the external sensor. Three heat transfer issues complicate this type of measurement:

- One issue is adequate contact between the sensor and the wall of the copper tube. All of the devices have some sort of clamping system to assure a close fit, but local variations in the copper surface as well as dirt accumulation or oxidation on the surface can add a contact resistance. This resistance leads to a higher temperature reading for the suction line than might otherwise occur. For some of the houses, we used heat sink compound with the Reference sensors. We found in the field, as well as in subsequent laboratory tests, that heat sink compound improved the temperature reading, but not significantly. Cleaning the heat sink compound off the lines after the test to avoid leaving an unsightly mess is time consuming. As a result, we do not recommend using heat sink compound for the superheat test.

- A more significant issue is whether the sensor is insulated. In particular, the Method 1 sensor has a high profile that makes insulating it difficult, but it already had a large plastic housing that provided some amount of insulation.

- Thermal mass of the temperature sensors is also an issue. Although all of the air conditioning systems operated for at least 15 minutes before each test (a requirement of the Method 3 software), we expect that the pressure and temperature of the refrigerant was still changing slightly even during the test (see Figure 3 for an example of this phenomena). Some of the temperature sensors, particularly the Method 1 sensor, had a very large mass. Such sensors have a slower response to temperature changes than a smaller sensor (e.g., the thermocouple used for Method 3).

### Target and Actual Superheat with Deviation above Target

Combined, the results from the preceding sections determine the target superheat, the actual superheat, and the deviation from the target superheat (Actual – Target). These results appear in Tables 6 and 7. Method 1 results are excluded from Tables 6a and 6b because this method does not provide a target superheat.

**Table 6a: "As Found" Target and Actual Superheats**

| Method | Target Superheat [°F (°C)] | | | | Actual Superheat [°F (°C)] | | | |
|---|---|---|---|---|---|---|---|---|
| | Site A | Site B | Site C | Site D | Site A | Site B | Site C | Site D |
| Ref. | 8 (4) | 9 (5) | 11 (6) | 14 (8) | 39 (22) | 47 (26) | 52 (29) | 46 (26) |
| 1[1] | - | - | - | - | 38 (21) | 40 (22) | 57 (32) | 43 (24) |
| 2 | 5 (3) | 7 (4) | 9 (5) | 11 (6) | 30 (17) | 45 (25) | 115[2] | 47 (26) |
| 3 | 10 (6) | 11 (6) | NA[3] | - | 38 (21) | 41 (23) | NA[3] | - |

[1]Method 1 does not calculate a target superheat
[2]This value is erroneous and was traced to a computer error.
[3]Either operator error or a computer error caused the Method 3 data to be missing for Site C as found.

**Table 6b: "Post Charging" Target and Actual Superheats**

| Method | Target Superheat [°F (°C)] | | | | Actual Superheat [°F (°C)] | | | |
|---|---|---|---|---|---|---|---|---|
| | Site A | Site B | Site C | Site D | Site A | Site B | Site C | Site D |
| Ref. | 8 (4) | 5 (3) | 10 (6) | 10 (6) | 8 (4) | 5 (3) | 9 (5) | 10 (6) |
| 1[1] | - | - | - | - | 8 (4) | 7 (4) | 13 (7) | 8 (4) |
| 2 | 5 (3) | 5 (3) | NA[2] | 10 (6) | 4 (2) | 7 (4) | NA[2] | 8 (4) |
| 3 | 7 (4) | NA[3] | 10 (6) | - | 8 (4) | NA[3] | 9 (5) | - |

[1]Method 1 does not calculate a target superheat
[2]A computer error prevented the Method 2 from saving the data for Site C post-charging.
[3]Either operator error or a computer error caused the Method 3 data to be missing for Site B post-charging.

## DISCUSSION

Based on our results, we recommend that Method 1 not be used for superheat tests of residential cooling systems at this time. One reason is that Method 1 does not report a target superheat. Instead, it reports a qualitative text string related to charge deviation: high, low, or acceptable (denoted as Hi, Lo, or Ok, with a modifying plus or minus sign to indicate whether it is a little bit high or low), or N/A, which means that the input data that Method 1 uses to calculate the target are outside an acceptable range. For this reason, the superheat target values exclude Method 1.

**Table 7a: "As Found" Deviation above Target Superheat**

| Method | Deviation above Target Superheat [°F (°C)] | | | |
|---|---|---|---|---|
| | Site A | Site B | Site C | Site D |
| Ref. | 31 (21) | 38 (21) | 41 (23) | 32 (18) |
| 1[1] | N/A | Hi++ | N/A | N/A |
| 2 | 25 (14) | 38 (21) | 106[2] | 36 (20) |
| 3 | 28 (16) | 30 (17) | NA[3] | - |

[1]Key to Method 1 results: N/A means Not Applicable/Out of Range, Hi++ means very high
[2]This value is erroneous and was traced to a computer error.
[3]Either operator error or a computer error caused the Method 3 data to be missing for Site C as found.

**Table 7b: "Post Charging" Deviation above Target Superheat**

| Method | Deviation above Target Superheat [°F (°C)] | | | |
|---|---|---|---|---|
| | Site A | Site B | Site C | Site D |
| Ref. | 0 (0) | 0 (0) | -1 (-1) | 0 (0) |
| 1[1] | Ok- | Lo | Lo | Ok- |
| 2 | -1 (-1) | 2 (1) | NA[1] | -2 (-1) |
| 3 | 1 (1) | NA[2] | -1 (-1) | - |

[1]Key to Method 1 results: OK- means the reading is fine, but a little low, Lo means that the reading is low.
[2]A computer error prevented the Method 2 data from being saved for Site C post-charging.
[3]Either operator error or a computer error caused the Method 3 data to be missing for Site B post-charging.

The use of text strings is not a problem by itself. However, for three of the four "as found" cases, Method 1 listed N/A in response to its actual superheat test, even though the systems (including the two brand new ones) were substantially undercharged. Furthermore, it reported other problems to be more important than the undercharge, so it did not give a complete diagnosis. This indicates that Method 1, when configured as when we tested, has very limited utility for assessing the refrigerant level in residential cooling systems. It is important to note that our analysis is limited to assessing charge levels. Method 1 might be very useful at locating important problems with air conditioning systems, but, without a measured entering wet bulb temperature, it does not seem appropriate for the specific problem of diagnosing refrigerant charge problems.

A more troubling problem with Method 1 is that its web data analysis sometimes differs from the PDA field analysis or its diagnoses are wrong. For example, all the cases that the PDA listed as N/A in the field were later diagnosed as undercharged by the web data analysis. Furthermore, at Sites B and C post-charging, Method 1 indicated the charge level was "Lo", which is contrary to the correctly charged condition indicated by the other methods. Having a technician obtain a diagnosis in the field, repair the equipment according to that diagnosis, and then find out later that there was a different diagnosis that may eliminate the need for the repair or worse, indicate that the repair should not have been performed, is a severe shortcoming. The "mutating" diagnosis problem is probably easy to rectify by better coordinating the diagnoses of the PDA software and the web site. However, the incorrect indication of charge level may be more difficult to rectify. Until these problems are solved, the method is unreliable for assessing charge levels in residential cooling systems.

Regarding Methods 2 and 3, uncertainties in the measurements lead to variations in target superheat, actual superheat, and superheat deviation between methods. These variations are as much as 5°F (3°C) for target superheat (Site A as found), 9°F (5°C) for actual superheat (Site A as found), and 8°F (4°C) for superheat deviation (Site B as found). As a comparison, laboratory test data from Farzad and O'Neal (1988) for capillary-tube-controlled equipment indicate a 10ºF (6ºC) error in superheat deviation can result in a charge assessment difference of about 5 to 9%, depending on outdoor temperature.

In spite of these variations, Methods 2 and 3 agree on their diagnoses and should result in similar actions to correct charge deficiencies. As a result, the variations described above may seem less significant. However, at all four sites in this study, the air conditioners were so undercharged (about 15 to 30%) that even the smallest "as found" deviation (25°F (14°C) for Method 2 at Site A) indicates a substantial problem, and the agreement in diagnoses should be expected. This raises a question of whether the methods would perform as well for air conditioners that were better charged. Examining the post-charging cases in Table 7b offers a partial answer. Where comparisons are possible, the post-charging superheat deviations never differ by more than 2°F (1°C), which suggests that Methods 2 and 3 will produce very similar results for correctly charged systems.

### Charge Effects on Equipment Performance

Table 8 shows the "as found" and "post-charging" total cooling capacities, energy efficiency ratios (EER), and power consumption that we measured or calculated, as well as the fractional changes in these parameters due to charging. A small amount of the changes in power draw, capacity, and EER can be attributed to small changes that occurred in ambient and outdoor temperatures between the "as found" and "post charging" conditions. Tables 3a and 3b list those temperatures.

**Table 8: Summary of Cooling Equipment Performance**

| | | Site A | Site B | Site C | Site D |
|---|---|---|---|---|---|
| Charge Added [oz (kg)] | | 15 (0.2) | 11 (0.3) | 20.5 (0.6) | 19 (0.5) |
| Fraction of Total | | 17% | 13% | 33% | Unknown |
| Air-Handler Airflow [cfm (L/s)] | | 1,240 (587) | 1,260 (593) | 1,320 (623) | 780 (368) |
| Capacity | "As Found" [tons (kW)] | 2.1 (7.4) | 2.2 (7.7) | 2.4 (8.4) | 1.7 (6.0) |
| | "Post Charging" [tons (kW)] | 2.8 (9.8) | 2.6 (9.1) | 3.3 (12) | 2.3 (8.1) |
| | Fractional Improvement | 33% | 18% | 38% | 35% |
| EER | "As Found" [Btu/Wh] | 5.2 | 7.0 | 8.8 | N/A |
| | "Post Charging" [Btu/Wh] | 6.2 | 7.5 | 10.6 | N/A |
| | Fractional Improvement | 19% | 7% | 20% | N/A |
| Total Power Draw Increase [Btu/h (W)] | | 1,840 (538) | 965 (283) | 1,495 (438) | N/A |
| Fractional Increase | | 11% | 7% | 13% | N/A |

The fractional charge increases listed in Table 8 are based on the total amount added divided by the factory charge listed by the manufacturer on the equipment rating plate. At Site D, this rating was not available. Service technicians should record on the rating plate the actual amount of charge that they have installed. Many rating plates have space explicitly available for this purpose. However, our field experience indicates that this is rarely done.

The capacities, efficiencies, and power draws in Table 8 are based on detailed measurements of air-handler airflow, supply and return temperatures and humidity, and power consumption made as part of the Reference measurements. We did not measure power consumption at Site D, because we could not safely connect our current and voltage sensors to the cooling equipment. As a result, we could not assess efficiency changes at Site D.

As expected, properly charging the cooling equipment significantly improved performance in terms of increasing capacity and efficiency. After charging, total cooling capacity improved by 18 to 38% and EER improved by 7 to 20%. Power consumption increased substantially after charging (increases of 280 to 540 W, or 7 to 13%). While this might seem to be a cause for concern to utilities in terms of peak electrical load, it is important to recognize that the increased capacity resulting from proper charging means that typically oversized cooling equipment is less likely to operate at its full-load. Proper charging thus increases the diversity of the aggregate air-conditioning-related load when the performance of many houses is considered together, which effectively reduces the utility peak.

An interesting result from this research is that two of these systems (Site C and Site D) were so undercharged that their vapor saturation temperatures were below 32°F (0°C). This temperature approximates the evaporator surface temperature. As a result, the evaporators iced up during the "as found" tests. We expect this greatly limited the airflow through the systems. Had we measured the airflows while the coils were iced up and included that effect in our capacity calculations, the "as found" capacities for Sites C and D would be even lower than those reported in Table 8. The systems in Sites B, C, and, particularly, D were already prone to icing even with proper charge due to their low airflow (314, 330, and 222 cfm/ton respectively).

## CONCLUSIONS

The importance of refrigerant charge to residential cooling performance is clear, as is the need to use a superheat test. Methods 2 and 3 correctly assess refrigerant charge levels. At this time, Method 1 seems inappropriate for assessing refrigerant charge levels of residential cooling equipment. Note that the reference method is too complicated and time intensive for a service technician to consider it as a practical alternative. However, this is not really a consideration, because it is intended only for research use.

There are problems with all of the methods, such as lost data for Methods 2 and 3, and some problems with deviations in pressure and temperature measurements. In the short term, diligence on the part of the service technician and the use of well-developed, reliable automation techniques seem to be the best solutions to these problems.

To address the significant performance degradation associated with refrigerant charge, we recommend that the building industry develop a standard method or methods to assess refrigerant charge. The results of this project suggest that the challenge will be to design a robust tool that works in most field conditions, rather than to measure the required quantities accurately enough. In particular, research is needed to develop a method of assessing refrigerant charge in cool weather. The utility of temporarily elevating indoor enthalpy also needs to be examined to extend the periods when the superheat method can be used to test cooling equipment in hot, dry climates. Ultimately, the performance of residential air conditioning systems would be dramatically improved by the development, application, and contractor training of a standard methodology to conduct refrigerant charge testing.

## REFERENCES


Blasnik, M., T. Downey, J. Proctor, and G. Peterson.1996. “Assessment of HVAC Installations in New Homes in APS Service Territory”. Proctor Engineering Group Report for Arizona Public Service Company.

Farzad, M. and D.L. O’Neal. 1988. “An Evaluation of Improper Refrigerant Charge on the Performance of a Split System Air Conditioner with Capillary Tube Expansion”. Texas A&M University, Energy Systems Laboratory Report ESL-TR-88/07-01. July.

Farzad M. and D.L. O'Neal. 1989. "An Evaluation of Improper Refrigerant Charge on the Performance of a Split System Air Conditioner with a Thermal Expansion Valve". Texas A&M Energy Systems Laboratory, ESL-TR-89/08-01.

O'Neal, D.L., C.J. Ramsey, and M. Farzad. 1989. "An Evaluation of the Effects of Refrigerant Charge on a Residential Central Air Conditioner with Orifice Expansion". Texas A&M Energy Systems Laboratory, ESL-PA-89/03-01.

Rodriguez, A.G. 1995. "Effect of Refrigerant Charge, Duct Leakage, and Evaporator Air Flow on the High Temperature Performance of Air Conditioners and Heat Pumps". Texas A&M Energy Systems Laboratory, ESL-TH-95/08-01.

Proctor, J. 1997. “Field Measurements of New Residential Air Conditioners in Phoenix, Arizona”. ASHRAE Transactions, Vol. 103, Part 1, 406-415.

Proctor, J. 1998. “Monitored In-Situ Performance of Residential Air-Conditioning Systems”. ASHRAE Transactions, Vol. 104. Part 1. 1833-1840.

Proctor, J. 2000. “AC Performance Associated with AB970”. Presentation to the California Energy Commission. November 28. http://www.energy.ca.gov/ab970_standards/documents/presentations/2000-11-28_PROCTOR.PPT.